Energy Research and Development Division
FINAL PROJECT REPORT

NATURAL GAS OPTIMIZED ADVANCED HEAVY DUTY ENGINE

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PREFACE

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Natural Gas-optimized Advanced Heavy-duty Engine is the final report for Gas Optimized Advanced Heavy Duty Engine Concept project (contract number PIR-08-046) conducted by Volvo Technology Corporation. The information from this project contributes to Energy Research and Development Division's Transportation Program.

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ABSTRACT

The High Pressure Direct Injection gas engine concept has been further optimized by evaluating the influence of engine hardware configurations, exhaust gas recirculation and engine control settings. The fuel efficiency for the optimized engine under real driving conditions was estimated to be six percent better than for a diesel reference engine, the diesel replacement is above 95 percent, and greenhouse gas emissions were lowered between 22-30 percent depending on the composition of the natural gas. Greenhouse gas savings can reach 90 percent if biogas was used instead of natural gas. Emissions of nitrogen oxides and soot from the optimized engine were similar to those from the diesel engine. Methane emissions for the optimized engine were low and the methane released during on-the-road driving conditions was below the Euro VI emission standards proposed limit of 0.5 gallons per kilowatt hour and could be considered sustainable.

Catalysts optimized for methane oxidation were successfully evaluated. Even with the best catalyst the exhaust gas temperature during real driving conditions was found to be too low for efficient methane oxidation. Additional work on advanced heat management is therefore recommended. The High Pressure Direct Injection technique was especially beneficial for efficient methane oxidation but was also a key enabler for other high efficiency gas engine concepts such as lean burn spark ignited engines and engines of the dual fuel type.

Keywords: natural gas, heavy duty engine, fuel efficiency, greenhouse gas, methane emissions

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EXECUTIVE SUMMARY

Introduction

Medium and heavy duty natural gas engines are operated in a premixed mode with either lean burn or stoichiometric gas mixtures. A stoichiometric process is one in which the fuel is burned completely. The combustion process is initiated by spark ignition and is characterized by premixed flame propagation typical for Otto (gasoline) engines. The largest engines have a displacement of approximately nine liters (L) with an upper power level of approximately 300 horsepower (hp). A problem for these engines is poor part load efficiency due to throttled operation, a weakness shared with gasoline engines. Another problem is that natural gas puts higher requirements on the ignition system and needs more energy from the spark compared to engines fueled with gasoline. The upper engine load is limited by knocking, despite the typically very high octane number of natural gas of 120 and the risk for knocking increases with engine size. Strategies using cooled exhaust gas recirculation (EGR) have the potential to improve part load efficiency as well as increase the load range. Alternative gas engine technologies are required for engine sizes in the 12-16 L displacement range (400-600 hp).

Project Purpose

The research reported here is in response to research, development, deployment and demonstration (RDD&D) needs identified by the California Energy Commission to improve the operating fuel efficiency of heavy-duty natural gas engines, improve the power output of natural gas engines, and lower the incremental cost of natural gas vehicles. VTEC has been working in partnership with Westport Innovations to evaluate and further develop the High Pressure Direct Injection (HPDI) technology that allows natural gas engines to operate at the same high efficiency and high power as today's heavy-duty diesel engines. The use of the HPDI technology addresses the first two research needs and the third on lower cost has been taken into account using an engine system approach and potentially benefiting from Volvo's global supply base.

The major project objectives were:

- Diesel engine performance in terms of drivability and efficiency.

- Replacement of 90 percent of crude oil-based diesel.

- Greenhouse gas emissions lowered by 20 percent.

- Meeting US14 emission regulations.

- Keeping methane emissions at sustainable levels.

Project Results

The HPDI heavy duty gas engine technology was based on direct in-cylinder injection of gaseous fuel providing conditions for mixing-limited combustion similar to the process in conventional diesel engines. The gas was supplied using a special high pressure gas injector. Small quantities of diesel fuel were injected in order to accomplish ignition. A major advantage compared to alternative gas engine techniques was that there was no knocking restriction on

the upper load limit and the power range could be similar to that of conventional diesel engines, making the technique attractive for heavy duty transport applications. Hydrocarbon (HC) emissions were relatively low due to more complete combustion. Small amounts of particulate matter were formed during combustion due to the non-premixed operation of the engine despite the low sooting tendency of natural gas. Emissions of nitrogen oxides (NO_x), were similar as those from the diesel engine and the project was carried out assuming the HPDI engine would be equipped with an aftertreatment system of the same type as that of the diesel engine, including both a NO_x catalyst and a particulate filter. An additional dedicated methane catalyst was also evaluated in this project for the HPDI engine.

The performance of the HPDI engine concept further optimized in the project delivered the same torque and power as the corresponding diesel engine. It was not possible to evaluate the transient response in the present project but based on experience from other activities the optimized engine was likely have similar response and drivability as the diesel engine. The fuel efficiency as measured in diesel equivalent units was estimated to be six percent better with the optimized engine compared to a reference gas engine and was also six percent better than the diesel reference.

The actual diesel replacement for real driving conditions was above 95 percent. Methane emissions were also low so the high diesel replacement led to a lowering of greenhouse gas emissions between 22-30 percent depending on the composition of the natural gas, the higher number corresponding to pure methane. Greenhouse gas savings could reach 90 percent if biogas was used instead of natural gas.

NO_x and soot emissions from the optimized engine were similar to those from the diesel engine. Exhaust gas temperatures and flows were also similar, so the same aftertreatment system as the diesel would meet the US14 emission regulations.

Methane emissions for the optimized engine were low and the methane released during on-the-road driving conditions was below the EUVI limit of 0.5 g/kWh (0.37 grams per brake horsepower-hour) and could be considered sustainable. Methane greenhouse gas emissions would offset approximately five percent of the benefit obtained by replacing 95 percent of diesel with natural gas.

All major objectives were met and the project will substantially contribute to the introduction of heavy duty vehicles in the California market.

The project task to evaluate and select a catalyst optimized for methane oxidation was successfully completed. The exhaust gas temperature during real driving conditions was found to be too low for efficient methane oxidation with the best catalyst. Additional work on advanced heat management was therefore recommended. Efficient methane oxidation was especially important for the HPDI technique but was also a key enabler for other high efficiency gas engine concepts such as lean burn spark ignited engines and engines of the dual fuel type.

The optimization performed in the present project concerned both hardware selection and the tuning of control settings. The engine was to a large degree still optimized for diesel

combustion and additional work was recommended to optimize the engine for the HPDI gas technology. Such work has the potential to lower fuel consumption even further and to decrease engine out emissions; the latter activity providing the potential to lower the complexity and the cost for the aftertreatment system.

Project Benefits

This project will substantially contribute to the introduction of heavy duty vehicles in the California market that use natural gas as fuel. Natural gas produces less carbon dioxide and NOx than conventional fuels used for vehicles, which can help reduce greenhouse gas emissions that contribute to climate change as well as other emissions that cause air pollution.

CHAPTER 1:
Engine Setup and Procedure

1.1   Setup

Experimental work was performed with a single-cylinder research engine at Volvo and a multi-cylinder production-type engine at Westport. The high flexibility and good control of a single-cylinder engine allow tests to be carried out varying parameters over large range and give the opportunity to detect also weak responses. A multi-cylinder engine is needed for final verifications of results including transient performance but also for defining proper boundary conditions for the single-cylinder engine. Data characterizing the engines used in the project are given in Table 1.

Table 1: Parameters Characterizing the Engines Used in the Project.

Parameters	Single-cylinder	Multi-cylinder
bore, mm	131	131
swept volume, dm^3	2.1	2.1
number of cylinders	1	6
swirl	0-6	quiescent
stroke, mm	158	158
compression ratio	16-18	17
EGR	yes	no
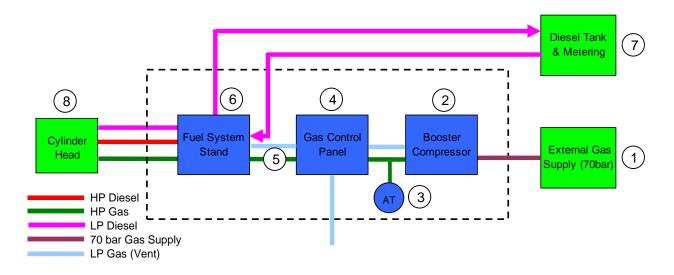
The picture in Figure 1 shows the single-cylinder test cell with the engine in front, the engine brake to the left and the special HPDI boost compressor to the right.

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Figure 1: Single-Cylinder HPDI Research Engine.

A schematic overview of the HPDI single-cylinder engine system is given in Figure 2. Three of the entities indicated in the figure (in blue color): Fuel System Stand, Gas Control Panel and Booster Compressor are unique HPDI components developed and manufactured by Westport. Other entities (in green) correspond to parts of the engine test cell that were modified by Volvo for gas engine operation.

Figure 2: Schematic Illustration of the HPDI Engine System.

Brief descriptions of major HPDI engine components are given in the following text items with numbers referring to those in Figure 2.

1   Natural gas at a pressure of 70 bar (1015.26 psi) is supplied to the engine lab from an external gas storage using a system designed for good functionality and highest degree of safety.

2   The booster compressor is a special lab device designed to increase the pressure of the gaseous fuel from the supply pressure to the desired engine operation pressure. In vehicle applications the high gas pressure is normally obtained by a pump in a tank containing liquefied natural gas (LNG).

3   The accumulator tank has the purpose is to smooth out pressure variations from the booster-compressor and is placed in a separate vented compartment to obtain maximum safety. The tank is a production component used on trucks fitted with the Westport HPDI system.

4   The gas control panel allows closing the gas flow to the engine and emptying high pressure gas from the system.

5   A Coriolis flow meter (Promass 63 from Endress+Hauser) measures the high pressure gaseous fuel flow supplied to the engine.

6   The fuel stand is used to regulate and control the diesel and gas pressures. A Common Rail pump supplies high pressure diesel fuel to a Fuel Conditioning Module (FCM), which uses a special regulator to maintain the gas pressure slightly below the diesel pressure, and an Inlet Metering Valve controls the diesel pressure as required.

7   Diesel fuel is supplied to a low pressure transfer pump which feeds the Common Rail pump. Minor modifications of the standard system were required for the diesel flow metering.

8   The HPDI injector, Figure 3, was specially designed by Westport to fit the Volvo engine with certain adaptations needed to the engine head for the supply of high pressure gaseous fuel and diesel.
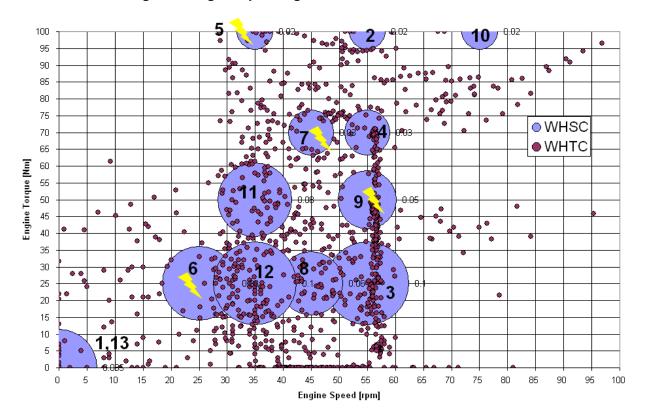
9   A special HPDI control system is interfaced with the engine control system including crank angle synchronization, timing and fuel demand. The combined control system also forms an essential part of the safety system.

1.2 Procedure

The typical procedure for obtaining data from the single-cylinder engine is to set all relevant engine parameters to fixed values and let the engine stabilize for approximately five minutes. Once a stable condition is reached data is collected for approximately two minutes. Cylinder pressure data from 30 cycles are sampled during a short period, in the course of the two minutes measurement interval. In addition to the standard test lab equipment an FTIR is used to measure methane and other non-regulated gas species.

Engine operating conditions were selected from the World Harmonized Stationary Cycle (WHSC). Figure 4 shows all operating points in the WHSC with the weight factor for each point illustrated by the size of a blue circle. Also the points in the World Harmonized Transient Cycle (WHTC) are included in Figure 4 shown as red dots.

Figure 4: Engine Operating Conditions for WHSC and WHTC.

CHAPTER 2:
Combustion Concept Development

The HPDI combustion concept was evaluated and further improved in several stages of the project, described in the following subsections.

2.1　State of the art HPDI engine characterization

In a first stage, tests were performed using an initial set of hardware and software parameters to create a reference data base representing the engine performance at the start of the project. Developments of the combustion system made in the project were subsequently evaluated by comparing engine performance with this reference data. The initial engine tests also had the important purpose to evaluate to which degree the single cylinder engine is able to show results relevant to a production-type multi cylinder engine. The reference set of hardware and software parameters corresponding to a state of the art HPDI multi-cylinder engine were made available to the project by Westport.

Five operation conditions from the WHSC, with load and speed given in Table 2, were used for the purpose of comparing the multi- and the single-cylinder engines.

Table 2: Operation Conditions for the Selected WHSC Points
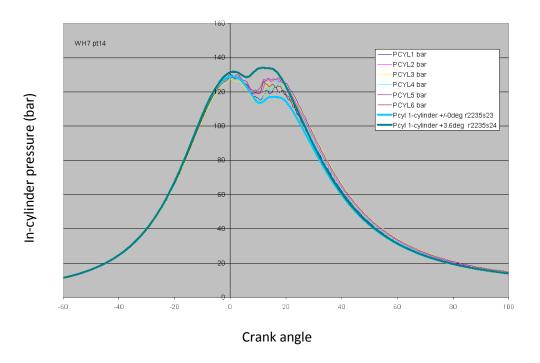
Operating point	load, Nm	speed, rpm
WH4	1790	1287
WH5	2350	1037
WH7	1810	1125
WH9	1282	1287
WH10	2242	1524
WH12	628	1037

2.1.1　In-cylinder pressure traces

Figure 5 shows measured in-cylinder pressure traces from the single cylinder and the multi cylinder engines for the WHSC 7 operating condition. It can be observed that the pressure traces differ slightly between the individual cylinders of the multi-cylinder. The pressure trace from the single cylinder engine corresponds to one case with early SOI (green curve) and one case with late SOI (blue curve) and by varying the SOI, the single cylinder can mimic any of the cylinders in the multi cylinder engine.
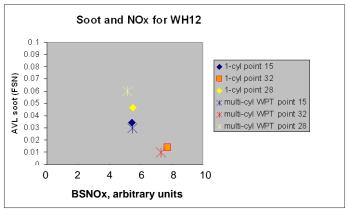
Figure 5: Measured in-Cylinder Pressure Traces for WHSC 7. Thick Lines Correspond to Single Cylinder Data and Thin Lines to Multi Cylinder Data.

Crank angle
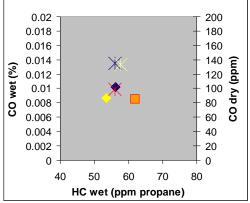
2.1.2 Exhaust gas emissions

Single-cylinder engine results are compared with multi-cylinder data in Figure 6 for three different parameter settings, all corresponding to the WH12 low load and low speed condition. The left side of Figure 6 shows measured emissions in form of soot and NO_x from the two engines. Data from the single- and the multi-cylinder engines are presented by solid symbols and stars respectively. Data from different points are distinguished by the colors of the symbols. For two of the points, 15 and 32, there is an almost perfect match between the results from the two engines. For the third point, 28, the single-cylinder gives slightly lower soot emissions than the multi-cylinder. The right side of Figure 6 compares emissions of CO and total HC from the two engines. Emissions of HC are similar but the single cylinder generally produces slightly less CO. The overall conclusion is that the single-cylinder reproduces the emission performance of the multi-cylinder with sufficient accuracy for WH12 operating condition.

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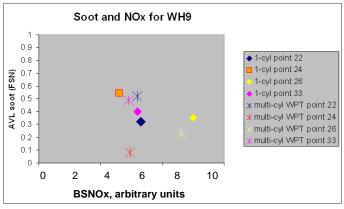
Figure 6: Comparison of Emissions Measured in the Single- and the Multi-Cylinder Engines, Soot and NO$_x$ to the Left and CO and Total HC to the Right Side of the Figure.
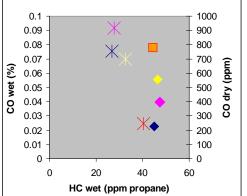
The WH9 operating condition corresponds to medium load and medium speed conditions. Four points were investigated in the single-cylinder engine for WH9 and a comparison of results is given in Figure 7. As shown in the Figure, the NO$_x$ emissions are very similar also for this condition. Soot emissions differ to a larger extent. It can be noted, however, that the multi-cylinder soot measurement for point 24 is an outlier and if point 24 is disregarded the agreement becomes good also for soot emissions. Emissions of CO and HC from the two engines, on the right side of Figure 7, differ to a larger degree. Depending on the point, CO emissions are either higher or lower in the single-cylinder whereas HC emissions are generally higher than in the multi-cylinder engine. If, again, point 24 is disregarding there is a clear trend with lower CO emissions in the single-cylinder. For WH9 the difference in emissions is larger than is satisfactory, the reason being an unidentified difference in the set-up of the engines. Despite observed differences between the single- and multi-cylinder engines for the WH9 operating condition, continued measurements were judged meaningful, but care should be taken before drawing final conclusions.
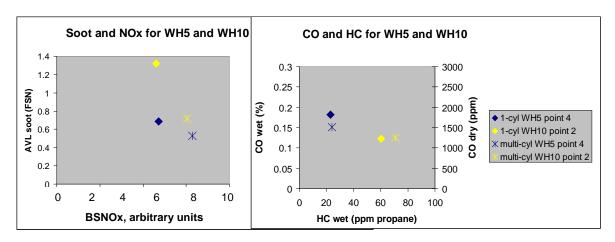
Figure 7: Comparison of Emissions Measured in the Single- and the Multi-Cylinder Engines, Soot and NOx to the Left and CO and Total HC to the Right Side of the Figure.

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WH5 and WH10 are both high load operating conditions, WH5 with medium speed and WH10 high speed conditions. Due to initial problems running the high load cases, only a few points were investigated in the single-cylinder engine. Results presented in Figure 8 indicate that, also at high load, the emission performance of the single cylinder is reasonably close to that of the multi cylinder engine.

Figure 8: Comparison of Soot, NOx CO and Total HC Emissions Measured in the Single- and the Multi-Cylinder Engines.

The general conclusion from all tested conditions is that the single-cylinder engine is able to reproduce results from a production-type multi-cylinder engine with a sufficient degree of accuracy to guide further development.
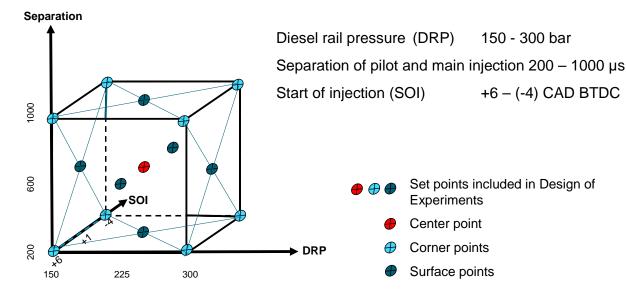
2.2 Engine hardware evaluation

The next objective was to improve the HPDI gas engine combustion concept by evaluating the hardware parameters considered most important: swirl and compression ratio (the compression ratio was varied by using two different pistons with similar shape). Improvements of engine performance were evaluated by comparing with previous reference engine data.

2.2.1 Design of Experiments

The investigation of the part load conditions was performed by means of a Design of Experiments (DoE) procedure which comprised three parameters (injection pressure, diesel gas separation and start of gas injection) and 17 measurements in a Central Composite Faced (CCF)-type of design, see Figure 9.

Figure 9: Graphic Presentation of CCF Design of Experiments in Operating Point WH12.

Diesel rail pressure (DRP)   150 - 300 bar

Separation of pilot and main injection 200 – 1000 µs

Start of injection (SOI)   +6 – (-4) CAD BTDC

Inlet and exhaust manifold pressure, as well as inlet manifold temperature and diesel pulse width were kept the same as for the reference engine, see Table 3. The gas pulse widths were chosen such as to keep the measured single cylinder torque at a constant level in all 17 measurements. Variation of swirl caused a variation in volumetric efficiency and hence in the air flow rate. In order to keep air flow and lambda at the same level when swirl was varied, a small adjustment of the inlet pressure was made.

Table 3: Constant Settings for all DoE-Points in WH4 and WH12.

Parameter	WH4	W12	unit
Speed	1285	1030	rpm
Diesel pulse width	500	330	µs
Inlet manifold pressure	277	128	kPa
Exhaust manifold pressure	267	128	kPa
Charge temperature	43	45	°C

Table 4 lists the tested hardware configurations including variation of compression ratio, together with piston geometry, and swirl ratio. Four different configurations were investigated, the baseline configuration with high compression ratio piston with low swirl and the low compression ratio piston with three different swirl levels. Swirl was varied by means of swirl plates mounted at the outlet of the inlet ports of the engine.
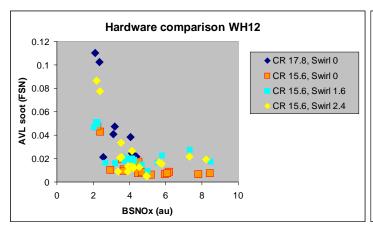
Table 4: Hardware Configurations.

Hardware configuration no	Compression ratio	Swirl no
1 (reference)	17.8	quiescent
2	15.6	quiescent
3	15.6	1.6
4	15.6	2.4

2.2.2 Results from measurements

Some initial observations can be made by studying Figure 19 and Figure 11 showing all measured data for operating condition WH12. The reference baseline configuration, 1, gives highest soot values but lowest BSFC. The combination 2, with low compression ratio (CR) and low swirl, gives lowest soot but highest methane emissions. Brake specific fuel consumption (BSFC) increases significantly for all specifications below a certain NO_x level. CO emissions stay below 200 parts per million (ppm) except for a few points where a substantial increase can be seen.

Figure 10: Emissions and Fuel Consumption for Four Hardware Combinations, $BSNO_x$-Soot to the Left and $BSNO_x$ BSFC to the Right Side of the Figure.

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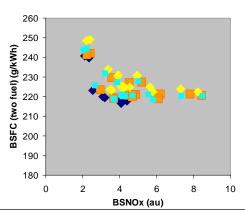
Figure 11: Emissions for Four Hardware Combinations, BSNO$_x$-CH$_4$ to the Left and CH$_4$-CO to the Right Side of the Figure.
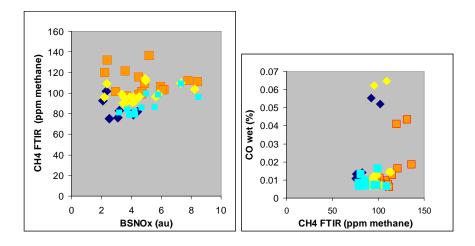
Figure 12 and Figure 13 show results from a continuous model created by interpolation of the DoE points for the reference configuration and configuration 2 respectively. In fact, all engine configurations show similar performance trends while varying rail pressure, separation, and start of injection (SOI). Regarding BSFC, the most influencing parameter is SOI while separation hardly has any impact. Higher rail pressures tend to improve fuel consumption. The most important parameter for BSNO$_x$ is the gas rail pressure, where higher pressures tend to increase Brake Specific Nitrogen Oxides (BSNO$_x$). The pilot to gas separation has an influence of similar magnitude to the timing variations. BS$_{soot}$ trends are similar to those for BSFC, with a large influence from SOI, especially for late timings. Gas rail pressure is almost as important as timing. The influence from pilot to gas separation is not significant.

Figure 12: Results for Operating Condition WH12 for the Reference Engine Configuration. Brake Specific Fuel Consumption (BSFC), NO$_x$ (BSNO$_x$) and Soot (BS$_{soot}$) are Plotted as Function of Rail Pressure (bar), Pilot Main Separation (us) and SOI (Start of Injection, CA BTDC). Blue and Green Lines Show 95% Confidence Level.
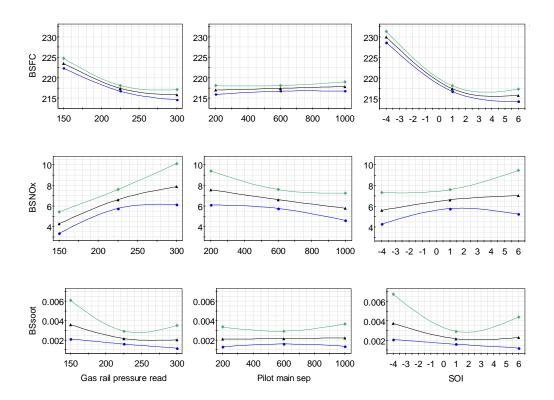
Figure 13: Results for Operating Condition WH12 for Engine Configuration 2. Brake Specific Fuel Consumption (BSFC), NO$_x$ (BSNO$_x$) and Soot (BS$_{soot}$) are Plotted as Function of Rail Pressure (Bar), Pilot Main Separation (us) and SOI (Start of Injection, CA BTDC). Blue and Green Lines Show 95% Confidence Level.
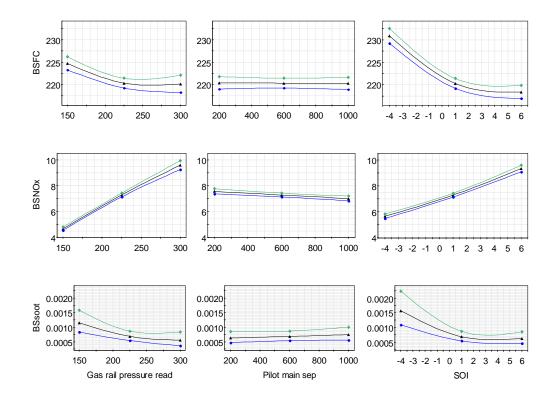
Figure 14 and Figure 15 show all measured data for the part load WH4 operating condition with similar overall pattern as for load point WH12. Lowest soot emissions are obtained for configuration 2 whereas configuration 4 gives the highest emissions of soot. There is a clear trend for soot, fuel consumption, and CO to increase with swirl. Engine configuration 2 has similar fuel consumption as the reference engine. A potential benefit with high swirl, to lower methane emissions, is shown in Figure 15.

Figure 14: Measured Results From the Engine for Four Hardware Configurations, BSNO$_x$ vs. Soot to the Left and BSNO$_x$ vs. BSFC to the Right Side of the Figure.
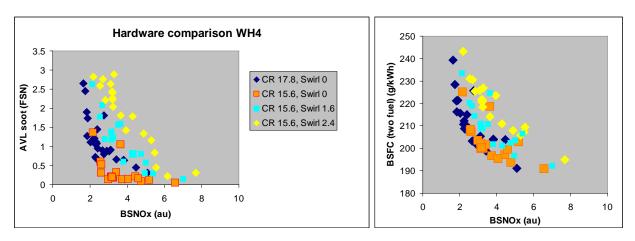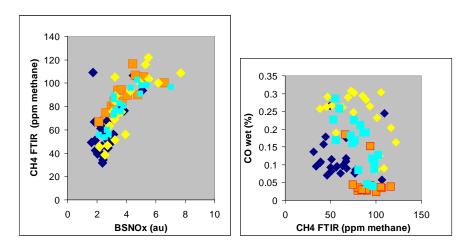
Figure 15: Measured Results From the Engine for Four Hardware Configurations, BSNO$_x$ vs. CH$_4$ to the Left and CH$_4$ vs. CO to the Right Side of the Figure.

In Figure 16, crank angle resolved data for three engine configurations are compared for operating condition WH4. The reference configuration (turquoise line) gives lower rate of heat release which can be explained by the higher pressure and hence the higher density in the cylinder. Comparing low and high swirl (blue and red line) it is apparent that combustion is faster with high swirl. However, despite the faster combustion the measured fuel consumption is higher. Higher swirl leads to increased heat losses that seem to have a dominating effect on fuel consumption.
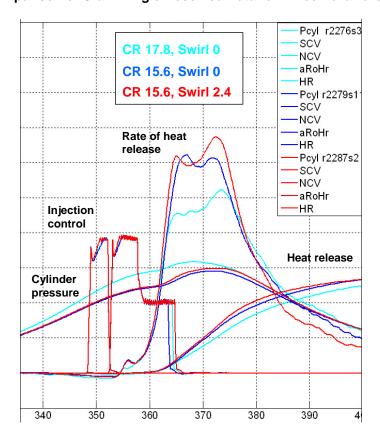
Figure 16: Comparison of Crank Angle Resolved Data for Three Hardware Configurations.

2.2.3 Results from Optimization

Based on measured data, models were developed and used to find the optimized settings for each hardware configuration at a common given NO$_x$ emission level. Optimized results for operating condition WH12 are shown in Figure 17 and indicate that the lowest fuel consumption is obtained with the reference engine. Engine configuration 2 gives only marginally higher fuel consumption but substantially lower emissions of soot but methane emissions are 30 percent higher than those from the reference engine.
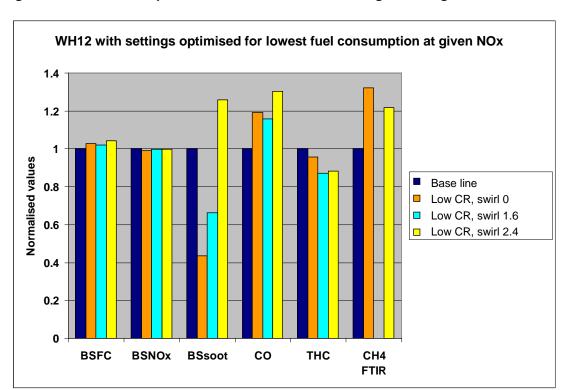
Figure 17: Normalized Optimized BSFC for the Different Engine Configurations for WH12.

Results from fuel consumption optimization for operating condition WH4 are shown in Figure 18. At this operating condition, configuration 2 shows the best performance for fuel consumption as well as for all emissions.
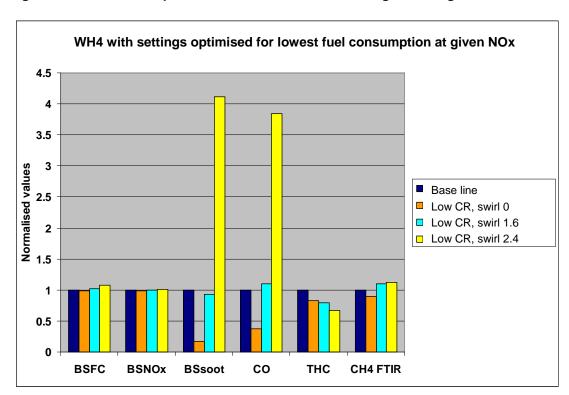
Figure 18: Normalized Optimized BSFC for the Different Engine Configurations for WH4.

Based on the fact that configuration 2, with low CR and low swirl, showed the overall best performance for emissions and fuel efficiency, and in addition allows highest torque for given in-cylinder maximum pressure and exhaust temperature limits, this hardware combination was selected for further investigation in the project.

2.3   EGR evaluation
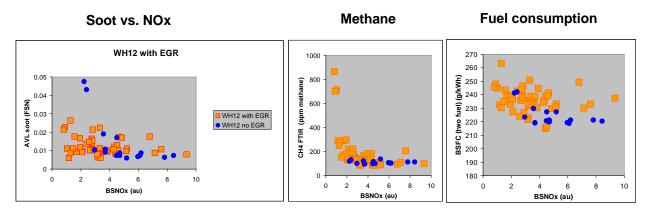
In a following step the HPDI gas engine combustion concept was further evaluated by investigating the effect of EGR with the most promising hardware combination, i.e. configuration number 2. The expected potential benefit with EGR is improved fuel efficiency by allowing injection timings and rail pressures optimized for lowest fuel consumption while limiting NO$_x$ emissions to target values by EGR.

2.3.1   Results from measurements

Figure 19 shows brake specific fuel consumption and engine out emissions of soot and methane as a function of measured NO$_x$ emissions for the WH12 operating condition. Data from previous non-EGR measurements (blue dots) are given in the same plots for reference. The left diagram in Figure 19 indicates that substantially better soot vs. NO$_x$ trade-off curves can be obtained with EGR. In fact, very low NO$_x$ can be reached without any substantial soot penalty. The middle diagram suggests a methane-NO$_x$ trade-off that is similar with EGR as without. The lowest NO$_x$ levels can only be reached with EGR, but with a large increase of methane

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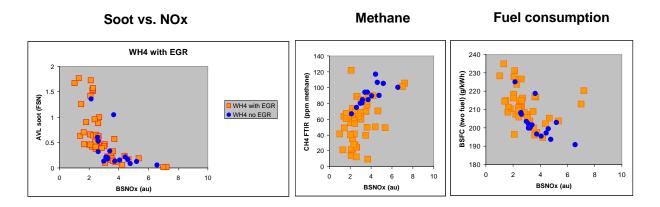
emissions. The results given in diagram to the right are not fully conclusive but indicate similar fuel consumption-NO$_x$ trade-offs with and without EGR.

Figure 19: Emissions and Fuel Consumption as a Function of NOx for WH12. Results Obtained with EGR are Marked in Orange and Non-EGR Results in Blue.

Soot vs. NOx　　　Methane　　　Fuel consumption

Results from all measured points for the WH4 operating condition are plotted in Figure 20. Again, studying the left plot, it is clear that a better soot-NO$_x$ trade-off curve is obtained with EGR. Concerning methane emissions, middle plot, the results are very interesting. There is no trade-off between methane and NO$_x$, instead methane and NO$_x$ decrease simultaneously both with and without EGR, however, EGR seems to have a large potential to lower methane emissions. The penalty going to low methane emissions is illustrated in the right plot showing substantial increase in fuel consumption for the settings with lowest NO$_x$ emissions, also having lowest emissions of methane. The exhaust gas temperature is very likely correlated with low methane emissions but high exhaust gas temperature can also indicate poor efficiency and high fuel consumption. The fuel consumption-NO$_x$ trade-offs seem, however, similar with and without EGR.

Figure 20: Emissions and Fuel Consumption as a Function of NO$_x$ for WH4. Results Obtained With EGR are Marked in Orange and Non-EGR Results in Blue.

Soot vs. NOx　　　Methane　　　Fuel consumption
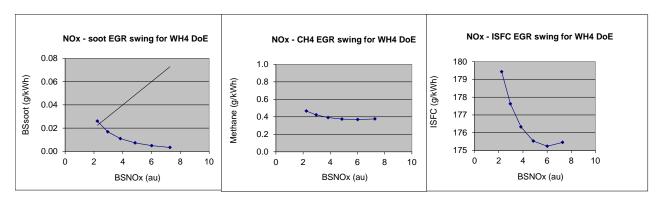
2.3.2 Results from Optimization

The overall objective was to evaluate potential benefits with EGR using results obtained with no EGR as reference. However, the comparison of fuel efficiencies between engines with and without EGR is non-trivial. Typically these [explain: single-cylinder and six-cylinder] engines have different air charging systems and different gas exchange losses. For a single-cylinder engine, gas exchange losses cannot be measured directly but are considered by setting proper engine intake and exhaust gas boundary conditions. The procedure used later in the project was to use gas exchange models for the complete systems corresponding to the two engines and use a combination of measured data and calculated results to eventually obtain final values for brake specific fuel consumption.

To obtain preliminary results DoE models were developed based on engine measured data for brake specific emissions and indicated, rather than brake specific fuel consumption. Figure 21 shows results from the DoE model for operating condition WH4 where emissions of soot and methane are plotted as function of NO_x emissions, with NO_x controlled by the amount of EGR fed to the engine. The plot to the right in Figure 21 shows indicated fuel consumption, ISFC, for different EGR levels.

Figure 21: Trade-Off Curves for NO_x vs. Soot (left), Methane Emissions (middle) and Indicated Fuel Consumption (Right).

The DoE model was used to further optimize settings with respect to the lowest specific fuel consumption given certain limits for exhaust gas emissions. Two sets of emission limits were selected corresponding to future regulations in the US and in Europe, see Table 5.
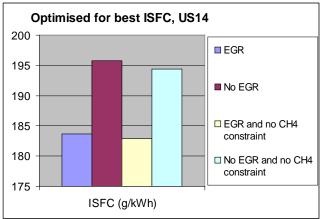
Table 5: Limits on Engine Out Emissions Used as Boundary Conditions While Optimizing Fuel Efficiency.
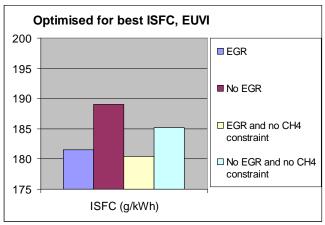
Emission regulation	NO_x	CH_4	soot
US14	NO_x-level US14	0.5 g/kWh	< NO_x/100
EUVI	1.5 * NO_x-level US14	1.0 g/kWh	< NO_x/100

[needs discussion to clarify engine out vs. tailpipe out since quoted regulations refer to tailpipe out.][EUVI, or Euro 6, is the lower levels being phased in by 2016. US14 refers to 2014 standards that add GHG control to US 2010 levels.]

Results optimized for best indicated fuel efficiency for the WH12 operating condition are shown in Figure 22. The lower engine out NO_x emissions for the US14 engine results in higher fuel consumption compared to the EUVI engine.[this seems to assume constant aftertreatment efficiency, but high aftertreatment efficiency may easily overcome moderately higher engine out emissions to still satisfy these regulations!! QUESTION: can you indicate what trade-offs would be needed to meet a NOx regulation of half or lower than the 2010/2014 level??] Comparing fuel consumption with and without EGR for the same emission limits, it is obvious that EGR gives opportunities to lower ISFC. The US14 EGR engine has ~13 g/kWh lower ISFC than the non-EGR version, shown in the left plot of Figure 22. The non-EGR engine is expected have lower gas exchange work but the difference is sufficiently large to expect a benefit also for the brake specific, BSFC, numbers. For the EUVI calibration, with its higher engine out NO_x, the differences in fuel consumption are smaller but still showing a ~7 g/kWh benefit using EGR. Also, as indicated in Figure 22, limits on methane emissions lead to increased fuel consumption for both the US10 and the EUVI engine, with and without EGR.
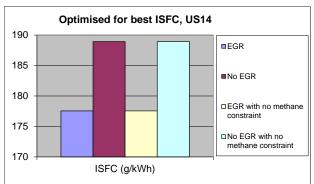
Figure 22: Indicated Fuel Consumption for the US14 and EUVI NO_x Limits, With and Without Constraints on CH_4 for the WH12 Operation Condition.
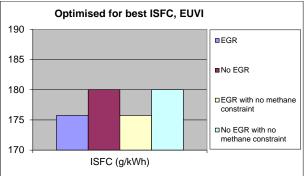
Results optimized for best indicated fuel efficiency for the WH4 operating condition is given in Figure 23. Without EGR the US14 engine consumes ~9 g/kWh more fuel whereas with EGR the fuel penalty is ~2 g/kWh. For the WH4 operating conditions limits on methane emissions are not limiting fuel efficiency, neither with nor without EGR, methane emissions being substantially lower than for the low load WH12 case. With EGR the US14 EGR engine has ~12 g/kWh lower ISFC than the non-EGR version, left plot of Figure 23. Also here, the difference is sufficiently large to expect a benefit also for the brake specific, BSFC, numbers. For EUVI the fuel consumption is ~4 g/kWh lower with EGR then without.

Figure 23: Indicated Fuel Consumption for the US14 and EUVI NO$_x$ Limits, With and Without Constraints on CH$_4$ for the WH4 Operation Condition.

CHAPTER 3:
Evaluation of Exhaust Aftertreatment

3.1   Evaluation of catalysts in synthetic gas bench

The catalytic activity of five methane oxidation catalysts was investigated in a synthetic gas bench (SGB) under conditions similar to those in engine exhaust. Table 6 lists the platinum-group metal (PGM) loadings and composition of the different catalysts.

Table 6: PGM Loadings of Samples.

Sample	PGM loading (g/ft³)	Pt:Pd	Pt (g/ft³)	Pd (g/ft3)
1a, 1b	250	1:5	42	208
2	125	1:4	25	100
3	125	1:4	25	100
4	120	1:1	60	60
5	300	0:1	0	300

Figure 24: Three Catalyst Samples.

Cylindrical samples with a diameter of 20 mm were drilled out of each catalyst monolith and the samples were then cut to a length of 40 mm, see Figure 24. All samples were initially stabilized in air for 3 hours at 600 °C. The catalytic activities for each sample were then

investigated in the SGB during temperature ramps from 110 °C to 670 °C with a temperature increase of 10 °C/min. The activities were also investigated for several different feed gas compositions but the background gas composition had only a minor influence on the methane oxidation capacity.

Figure 25: Methane Conversion From All Samples Investigated Using a Feed Gas Composition of 200 ppm CH$_4$,30 ppm HC, 200 ppm CO, 1200 ppm NO, 5% CO$_2$, 10% H$_2$O, and 10% O$_2$.

The methane conversion results from all investigated catalysts are shown in Figure 25. The conversion ranges between 66-95 percent at the highest temperatures. Sample 1, with high PGM loading and a Pt to Pd ratio of 1:5, shows the highest conversion of all samples at all temperatures. The difference between sample 1a and 1b was that the catalyst from which the samples were collected was expected to have an uneven distribution of PGMs and two samples from the same catalyst were investigated. Sample 1b generally performed slightly better than sample 1a.

Sample 4 with low PGM loading and a Pt to Pd ratio of 1:1, represents a typical "conventional" diesel oxidation catalyst (DOC). Compared to sample 1, representing a catalyst optimized for methane oxidation, the DOC has higher light-off temperature and lower maximal methane conversion.

Additional tests were carried out to investigate the CO and NO oxidation capacity of the catalysts. The lowest CO light-off temperature was observed with the catalysts with the highest Pt content and the highest light-off temperature was observed with the catalyst with only contained Pd and no Pt. The lowest NO oxidation capacity was observed for the sample with only Pd and no Pt.

3.2 Characterization of methane oxidation during engine tests

In a following step, two of the catalysts evaluated in the gas test bench, the methane catalyst (sample 1) and the standard DOC, were characterized during engine tests. Since the tests are

performed with the single cylinder engine, the size was chosen to correspond to 1/6 of a EUVI DOC (5.66? inch diameter and length 4 inch). The catalyst temperature is measured with a 1 mm thermocouple inserted in a channel in the center of the front end of the catalyst. The substrate is of standard type, NGK 400 CPSI.

The catalysts are mounted several meters downstream the engine and the exhaust tubing is insulated to keep a sufficient temperature. The engine is operated at stationary points chosen to cover the typical range of exhaust temperatures and flow rates for a catalyst positioned at a normal position close to the engine.

Figure 26 shows the methane conversion as function of catalyst temperature and flow rate for the methane catalyst. The figure shows 100 percent conversion obtained for temperatures above ~450 °C, which was to be expected although results from the SGB do not show complete conversion even at the highest temperatures. The results in Figure 26 also indicate that the sensitivity to the exhaust flow rate is low.

Figure 26: Methane Conversion as a Function of Temperature and Flow Rate for the Methane Catalyst. The Surface is a Polynomial Model Fitted to the Measured Points.
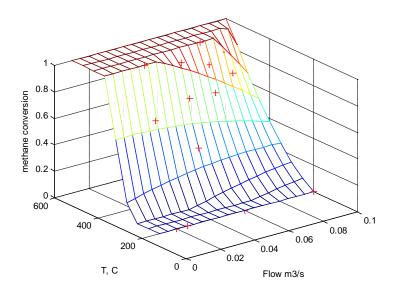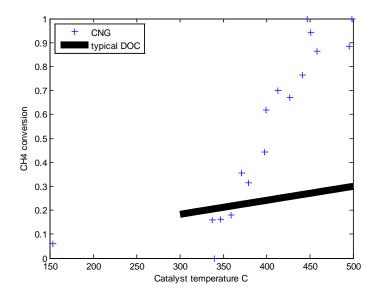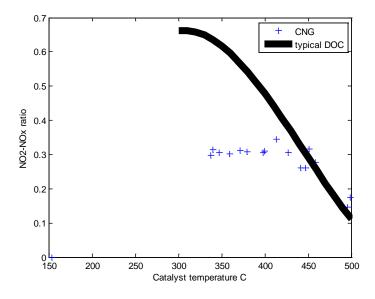
Figure 27 shows the measured conversion as function of temperature, both for the methane catalyst and for the DOC; results from the latter indicated by a broad black line. Based on these results it is obvious that a methane catalyst has a potential to efficiently reduce methane tail pipe emissions. The DOC, on the other side, does not perform well enough.

Also the oxidation of NO to NO_2 was studied for the two catalysts and the result is given in Figure 28. The DOC produces substantially more NO2 than the methane catalysts at temperatures below ~400 °C, indicating that the aftertreatment system may need to be re-optimized if a methane catalyst is added to the system.

Figure 28: Measured NO_2 to NO_x Ratio vs. Temperature for the Methane Catalyst, Blue + Marks, and the DOC, Broad black Line.

CHAPTER 4:
Evaluation of Production Engine Performance

Final evaluation of the optimized combustion concept and aftertreatment system ideally would be carried out on a multi-cylinder production type engine. In the present project only a non-EGR version of the multi-cylinder engine was available, so an alternative procedure was employed to evaluate a combination of experiments and simulations. The procedure can be summarized as follows:

1. Use non-EGR multi-cylinder reference engine results as a starting point

2. Develop and validate a gas exchange multi-cylinder engine model using data from the non-EGR multi-cylinder engine

3. Modify the gas exchange model to include a long-route EGR circuit used to calculated boundary conditions for single-cylinder engine corresponding to a multi-cylinder EGR engine

4. Optimize engine control settings for best fuel efficiency within limits for maximum in-cylinder pressure and exhaust gas temperature

5. Add EGR as needed to reduce NO_x to same engine out emission as the multi-cylinder non-EGR reference engine

6. Add optimized catalyst to reduce methane emission

The multi-cylinder engine model was developed using the GT-Power gas exchange software package. Model validation compared calculated results to measured data using the rate of heat release as input to the model. The model outputs inlet and exhaust gas temperatures and pressures as well as brake specific fuel consumption generated for eight stationary conditions characterized by loads and speeds given in Table 7.

Upon reaching good agreement with measured data, the model was extended to incorporate a long route EGR system and predict boundary conditions for a multi-cylinder EGR engine. The heat release rates for the EGR engine were assumed the same as those for the non-EGR engine, an assumption with good validity at least up to moderate amounts of EGR ($\leq$ ~20 percent).

Table 7: Multi-Cylinder Engine Operating Conditions

speed (rpm)	load (Nm)
570	1262
600	1021
600	1140
1152	1021
1200	1262
1716	1140
2322	1021
2418	1262

In a next step the optimization was performed by adjusting injection parameters such to obtain best fuel efficiency, disregarding possible increases of NO_x emissions but respecting limits of maximum in-cylinder pressure and exhaust gas temperature. Using proper multi-cylinder EGR engine boundary conditions, 22 load and speed points belonging to WHSC or to WHTC were selected to characterize the EGR engine performance. NO_x emission control was evaluated by varying EGR in three steps for each load speed point. Figure 29 shows brake efficiency for the engine with optimized settings relative to the results for the reference non-EGR engine as function of load, expressed as brake mean effective pressure (BMEP). Both for the lowest and the highest lowest the optimized engine reaches fuel efficiencies that are more than 8 percent better than for the reference engine. For each load three different efficiencies are shown, each corresponding to a given level of EGR. Typically, the higher the EGR level the lower the efficiency.

Not unexpectedly, soot emissions increase as EGR is added to the engine, but not dramatically. For a given engine-out emission of NO_x, the soot levels from the EGR and the non EGR engines are similar, and the soot-NO_x trade-off curves obtained by varying the injection timing for non-EGR engine are also similar to the ones obtained by varying the EGR level for the EGR engine using fixed injection parameter settings. Figure 30 shows soot-NO_x trade-off curves obtained by varying EGR for a range of operating points. Results for the non-EGR engine are indicated by filled symbols.

Figure 29: Improvement of Brake Fuel Efficiency for Different Loads, Filled Symbols Reference Engine and Hollow Symbols Optimized Engine.
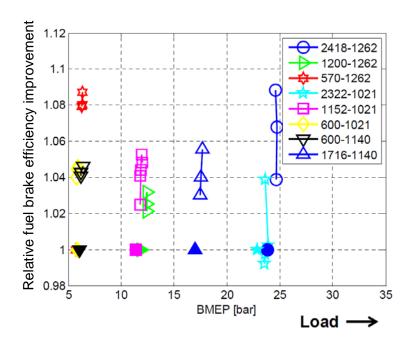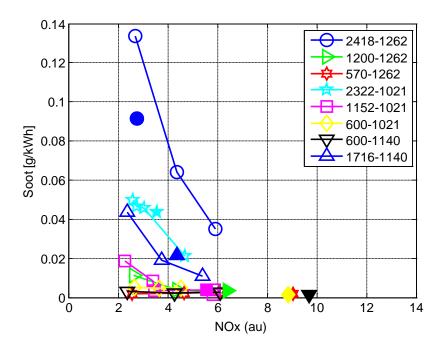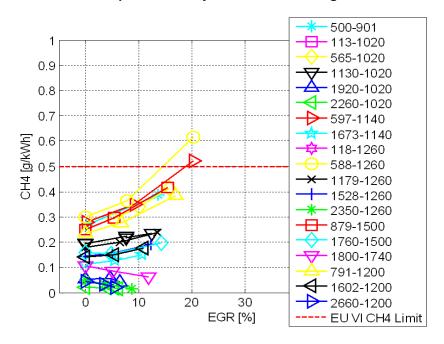

Figure 30: Soot vs. NOx Trade-Off for Different Operated Conditions, Filled Symbols Reference Engine and Hollow Symbols Optimized Engine.

An important project objective was to keep methane emissions at sustainable [levels and methane emissions might be expected to increase with increasing EGR. Figure 31 shows brake specific methane emissions as function of EGR and for a range of operating conditions. Indeed, for low and medium loads EGR has the effect to increase methane. However, for higher loads above 1800 Nm, methane emissions decrease as EGR increases, probably mainly caused by increased in-cylinder and exhaust gas temperatures leading to more complete methane oxidation. For the purpose of the project, sustainable methane emissions were defined as the EUVI limit, 0.5 g/kWh, indicated by the red dotted line in Figure 31. The figure shows that engine out emissions are substantially lower than the limit except for a few conditions with lowest load and highest EGR, indicating that special methane aftertreatment is not needed for the present engine.

Figure 31: Methane Emissions as Function of EGR for Different Operated Conditions With Load and Speed Given by Numbers in the Legend.

Another important project objective was to minimize the fraction of diesel. Figure 32 shows the diesel substitution ratio as function of load for the optimized and the reference engines. For the optimized engine the amount of diesel was selected as the minimum amount giving stable ignition and combustion. The resulting diesel substitution, calculated as substitution of diesel fuel energy, was between 93 and 98 percent, with the highest substitution at the highest load.

Figure 32: Diesel Substitution Ratio for Different Operated Conditions, Filled Symbols Reference Engine and Hollow Symbols Optimized Engine.
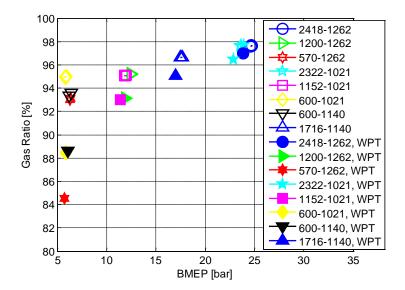
CHAPTER 5:
Virtual Vehicle Demonstration

Results from single-cylinder engine tests indicated a potential to reduce fuel consumption by optimize the engine settings. Optimized fuel efficiency led to increased engine out NO_x that, however, could be reduced to original levels by introducing EGR. As a final part of the project, fuel consumption and emissions from a vehicle was evaluated for real, on the road, driving conditions. This virtual vehicle demonstration was carried out using Volvo software simulating the performance of a Volvo VN 670 truck; see Figure 33, equipped with an HPDI gas engine optimized as described in the previous section.

Figure 33: Volvo VN 670 Truck.

5.1   Vehicle simulation methodology

The engine is modeled by simply mapping data from engine tests (fuel consumption, exhaust flow, exhaust temperature, engine out NO_x, and engine out CH_4) as function of EGR, load and speed, with values between measured points being obtained by interpolation using MATLAB. An example of such an engine map is given in Figure 34, showing the map for the exhaust gas temperature for zero EGR as function of engine speed and load. To avoid unrealistic values due to extrapolation, the nearest point is used for values outside the measured volume. During the drive cycle, i.e. at zero engine load, fuel consumption and emissions are set to zero and the exhaust temperature to 90 °C. The red + marks in Figure 34 indicate speed and load points for which engine measurements were available. Speed and load points during the actual drive cycle are shown as black + marks that form an almost vertical line, illustrating the typical condition for a vehicle aiming at driving at constant speed.

Figure 34: Example of engine map showing the exhaust gas temperature map. Speed and load points for measurements are shown as red + and the speed and load of the drive cycle is shown as black + (forming a vertical line).
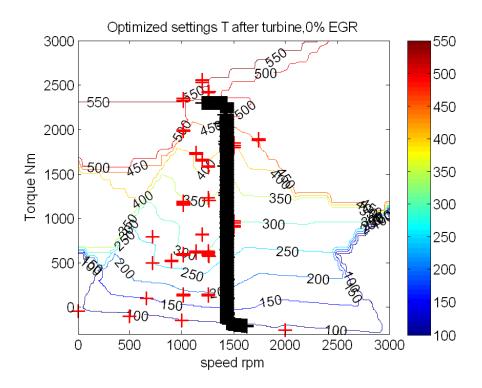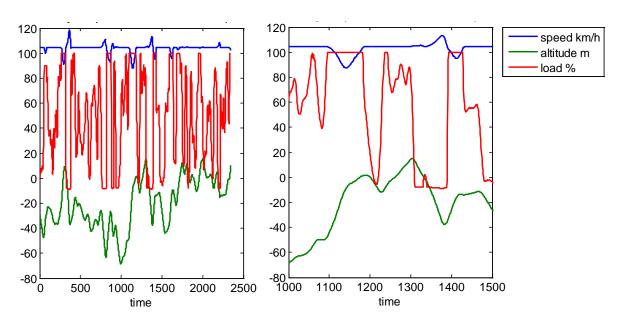
Table 8: Vehicle Configuration Used in the Rolling Hills Drive Cycle.

Engine	Power (kW)	Torque (Nm)	Tires	Final gear ratio	GCW (ton)	Front area (m²)	Cd
MD13 475	362	2304	275/70 R22.5	3.07	32.205	10.67	0.58

The vehicle part of the simulations program incorporates such factors as engine characteristics, total mass, air resistance, rolling resistance, gear ratio, etc. Data characterizing the truck used in the present work is given in Table 8. When evaluating the performance of the vehicle, the simulations program uses a description of a drive cycle including vehicle speed, altitude, temperature, etc. As is typical for long- haul driving conditions, the vehicle (or driver) is aiming for constant speed along the drive cycle. In the simulation, the speed is used as a set point value and the model is calculating speed and torque required to achieve the desired speed. The selected speed was 65 miles/h (~105 km/h) and simulations show that the vehicle is able to maintain the target speed with only minor deviations during the whole driving cycle (see Figure 35).

Figure 35: The Vehicle Speed, Altitude and Load From the Simulation of the Rolling Hills Cycle. The Figure to the Right is Showing the Complete Cycle Whereas the Left Figure Shows Part of the Cycle Between 1000 and 1500 s.

Outputs from the simulations where time resolved fuel consumption, engine out NO_x emissions and methane emissions.

Fuel consumption was calculated using a model based on interpolating experimental data to obtain a map similar to the temperature map displayed in Figure 34.

The tailpipe methane emissions were estimated based on a model of the optimized methane catalyst described in Chapter 3, the model consisting of a table of the methane conversion rates as a function of temperature and flow. Transient effects on methane emissions were captured in part by including the thermal mass of the exhaust system in the model. Input to the methane emission model was generated using separate maps of the engine out methane emission, exhaust gas temperature, and exhaust flow.

The on -road vehicle conditions were represented by the ”Rolling Hills Route” a 42 miles distance on I-77 in South Carolina between exit 73 – 27(see Figure 36). Complementary information is given by the green line in Figure 35 showing the altitude along the route.
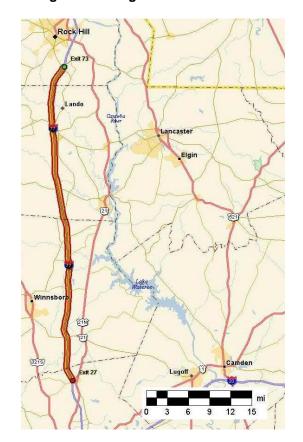
Figure 36: Map Showing the "Rolling Hills Route" on I-77 Between Exit 73 – 27.

5.2   Vehicle simulation results

The fuel consumed in the cycle for vehicles equipped with different engines is shown in Figure 37 to the left. With optimized engine settings and no EGR the fuel consumption is approximately 8 percent lower that the consumption with the reference engine (blue and turquoise line respectively). The fuel consumption is calculated as the diesel equivalent fuel consumption and Figure 37 presents the accumulated values in arbitrary units. A consequence of engine settings optimized for highest fuel efficiency is that NO_x emissions increase as described in Chapter 4. The accumulated NO_x emissions are given in the right side of Figure 37 showing approximately 11 percent higher emissions with the optimized engine compared to the reference. With an optimized engine and 2 percent EGR, the difference in NO_x emissions decreases to 6 percent(green curve in left plot) however also the difference in fuel efficiency decrease slightly as shown in the left plot.

Figure 37: Accumulated Diesel Equivalent Total Fuel Consumption, Left, and Engine Out NO$_x$ Emissions, Right, for Three Different Engines (Arbitrary Units).
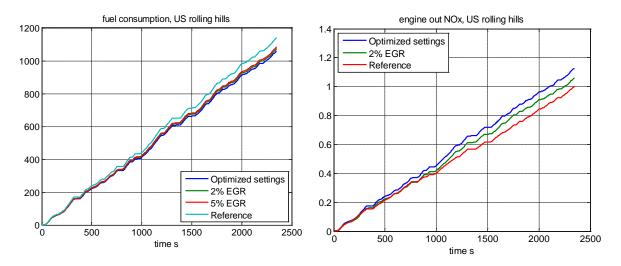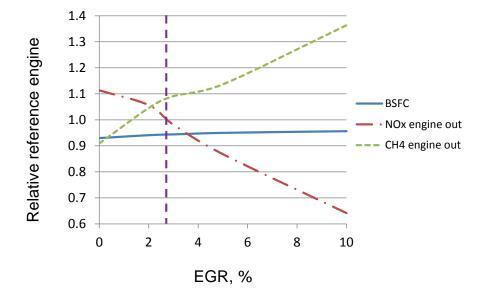
Figure 38 shows total fuel consumption and engine out emissions of NOx and methane as function of EGR. Numbers for fuel consumption and emissions in the figure are normalized relative the reference engine. Both NO$_x$ and methane emissions depends strongly on the level of EGR whereas fuel consumption is less sensitive.

Figure 38: Total Cycle Fuel Consumption and Emissions as Function of EGR
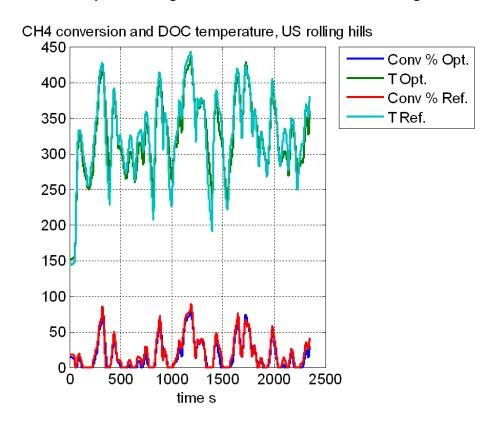
With approximately 2.5 percent EGR, vertical dashed line in Figure 38, the engine out NO$_x$ from the optimized and the reference engines are the same. Assuming same efficiency for the NO$_x$ aftertreatment system also tail pipe emissions will be the same, both engines meeting the US14 regulations. At the same NO$_x$ emission the fuel consumption with the optimized engine is 6 percent lower than the reference consumption, a significant improvement and a major result from the project.

5.3 Methane emissions

Figure 39 shows simulated results for methane catalyst conversion and temperature for the optimized engine with zero EGR and for the reference engine. Methane conversion rates are very similar for the two engines but clearly insufficient for substantial reduction of the total cycle methane emissions. The reason being that the average exhaust gas temperatures are too low for good performance, even with the optimized methane catalyst described in Chapter 3, see Figure 27.

Figure 39: Methane Conversion, Lower Curves, and Catalyst Temperature, Upper Curves, for the Optimized Engine With 0 EGR and the Reference Engine.

Resulting cycle average methane emissions are summarized in Table 9. Methane conversion might be considered surprisingly low, especially with a non-flat driving cycle. Fortunately, even with moderate EGR the HPDI engine out emission of methane is sufficiently low to meet also the EUVI stringent levels of 0.5 g/kWh and can thus be considered as sustainable. Other gas engine concepts do produce significantly higher methane emissions and do require methane aftertreatment. A possible enabler for such aftertreatment systems to become efficient is advanced heat management, a technique that was not investigated in the present project.

Table 9: Cycle Averaged Values Methane Emissions for Three Different Engines.

Methane averaged g/kWh	Optimized 0% EGR	Optimized 2% EGR	Reference
Engine out	0.20	0.23	0.22
System out	0.17	0.19	0.18
EATS efficiency	15%	17%	18%
EUVI legislation	0.50	0.50	0.50

CHAPTER 6:
Conclusions and Recommendations

6.1 Conclusions

The major project objectives were:

- Diesel engine performance in terms of drivability and efficiency

- Replacement of 90 percent of crude oil based diesel

- Greenhouse gas emissions lowered by 20 percent

- Meeting US14 emission regulations

- Keeping methane emissions at sustainable levels

All major objectives have been met.

Concerning performance, the optimized HPDI engine concept delivers the same torque and power as the corresponding diesel engine. It was not possible to evaluate the transient response in the present project, but based on experience from other activities the optimized engine is likely have similar response and drivability as the diesel engine. The fuel efficiency, measured in diesel equivalent units, is estimated to be 6 percent better with the optimised engine compared to the reference gas engine and, in fact, also 6 percent better than the diesel reference.

The actual diesel replacement for the "Rolling Hills" route was above 95 percent. Since methane emissions were low, the high diesel replacement leads to a lowering of greenhouse gas emissions between 22-30 percent depending on the composition of the natural gas; the higher number corresponding to pure methane. If biogas is used instead of natural gas, greenhouse gas savings can reach 90 percent.

Emissions of NO_x and soot from the optimized engine are similar to those from the diesel engine. Also exhaust gas temperatures and flows are similar and using the same aftertreatment system as the diesel the US14 emission regulations will be met.

Methane emissions for the optimized engine are low and the methane released during on-the-road driving conditions is below the EUVI limit and can be considered sustainable. The estimated methane greenhouse gas emission offsets approximately 5 percent of the benefit obtained by replacing 95 percent of diesel with natural gas.

6.2 Recommendations

The project task to evaluate and select a catalyst optimized for methane oxidation was successfully carried out. However, even with the best catalyst the exhaust gas temperature during real driving conditions was found to be too low for efficient methane oxidation. Therefore, additional work on advanced heat management is recommended. Although

absolutely necessary for the HPDI technique, efficient methane oxidation is a key enabler for other high efficiency gas engine concepts as lean burn spark ignited engines and engines of the dual fuel type.

The optimization performed in the present project concerned both the selection of hardware and the tuning of control settings. However, the engine is to a large degree optimized for diesel combustion and additional work is recommended to optimize the engine for the HPDI gas technology. Such work has a potential to lower fuel consumption even further and to decrease engine out emissions, the latter activity giving a potential to lower the complexity and the cost for the aftertreatment system.

APPENDIX A:
Acronyms

°C	degree Celsius
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
$BSNO_x$	Brake Specific Nitrogen Oxides
BS_{soot}	Brake Specific Soot
BTDC	Before Top Dead Centre
CA	Crank Angle
CCF	Central Composite Faced
CH_4	Methane
CO	Carbon Monoxide
CO_2	Carbon Dioxide
CR	Compression Ratio
DOC	Diesel Oxidation Catalyst
DoE	Design of Experiments
EATS	Exhaust Aftertreatment System
EGR	Exhaust Gas Recirculation
EUVI	Euro VI emission standards
FCM	Fuel Conditioning Module
FTIR	Fourier Transform Infrared Spectroscopy
GCW	Gross Combination Weight
HC	Hydrocarbons
HPDI	High Pressure Direct Injection
ISFC	Indicated Specific Fuel Consumption
LNG	Liquefied Natural Gas
NO	Nitrogen Monoxide
NO_2	Nitrogen Dioxide

NO$_x$	Nitrogen Oxides (NO + NO$_2$)
Pd	Palladium
PGM	Platinum-Group Metal
ppm	parts per million
Pt	Platinum
RDD&D	Research, Development, Demonstration and Deployment
rpm	revolutions per minute
SGB	Synthetic Gas Bench
SOI	Start Of Injection
T	Temperature
VTEC	Volvo Technology
US14	USEPA emission standards for 2014, adding GHG emissions to US10
WHSC	World Harmonized Stationary Cycle
WHTC	World Harmonized Transient Cycle